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The Secondary Flow in Compressor Blades at High Negative Incidences

By

W. D. Armstrong,
Cambridge University

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1956

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Introduction

Previous investigation has shown that when a non-uniform stream flows through a cascade of blades the resulting vortex system in the stream direction induces velocity components parallel and perpendicular to the blade span. Tests of compressor blades at low angles of incidence produce a downstream angle α_2 distribution of the general pattern between and within the blades shown in Fig. 1. This type of pattern is that expected for the vortex system shown in Fig. 2, where A is the distributed secondary vorticity due to the major turning of the stream within the blades and B is the vorticity arising from the trailing filament and shed circulations as defined in Ref. (1). Experiments and theoretical calculations(2), (3) using an impulse turbine cascade have demonstrated conclusively the nature of these flows. In a non-impulse cascade there is in addition a change of outlet angle which is caused by the reorientation of the streamlines to satisfy the continuity and constant pressure conditions. By using the concept of the actuator plane(4) this variation can be predicted. As has been shown(5) the actuator plane predicts a continuously increasing value of outlet angle α_2 for a compressor cascade as the cascade wall boundary layer is approached from the centre of the blade span.

In the investigation reported here a quite different distribution of outlet angle along the span was measured.

Experiments

The variation of outlet angle downstream of two cascades of compressor blades was measured along the span of the blade at different distances between the trailing edges of adjacent blades. A 150 H.P. pressure tunnel was used with 6 in. chord blades, having a span of 18 in. and a pitch of 6 in. The C4 blade profile camber was 40° and two inlet angles ($\alpha_1 = 45°$, $\alpha_1 = 50°$) were used, the corresponding incidences being $-13°$ and $-8°$. The upstream cascade wall boundary layer was approximately 1 in. thick and was of the normal turbulent form, in the Reynolds number range 1.1 to 1.4×10^5.

The results are shown in Fig. 3(a) and (b) and should be compared with the more conventional distribution of angle as in Fig. 1. To assist in the discussion the curves of Fig. 3(a) are repeated in staggered form in Fig. 4 and those of Fig. 3(b) in Fig. 5.

Additionally, during the experiments with $\alpha_1 = 45°$ the position of transition from laminar to turbulent flow on the blade suction surface was located by means of hot wire turbulent measurements+. No change in the position was found throughout the central 15 ins. of the blade span, it is therefore unlikely that the α_2 distribution is due to differences in Reynolds number along the blade.

Measurements/

+By a colleague D. L. Martlew

Measurements of the outlet angle have also been made using these blades, over a wide range of Reynolds number and a summary of the results is presented in Fig. 6.

Discussion

The following hypothesis is proposed to account for these experimental measurements.

At high negative incidences and low Reynolds numbers the deflection of the blading is small, e.g., approximately 11° in the first of these experiments. The secondary distributed vorticity will therefore not be very large. With an incidence of -13° however it can be expected that the stagnation point will be well round the nose on the convex blade surface. Consider now two streamlines near a stagnation streamline. As they approach the stagnation point, they divide, one passing round the convex profile and being turned a little more than 11° whilst the other, after the stagnation point, is turned rapidly over 90° in the opposite direction, followed by the main 11° turning in the blading direction. The main secondary distributed vorticity will therefore be heavily opposed near the concave surface and it might be expected that with high negative incidences this "reversed" vorticity component will predominate. It should be noted that this additional vorticity is not to be confused with the nose vortices discussed previously[5]. These are of the opposite sign since they are caused by the initial reverse rotation just upstream of the stagnation point.

If there is then a strong "reversed secondary vortex" lying in the corner of the wall and the concave blade surface the air outlet angle would first decrease and then increase rapidly as the wall is approached from the blade centre line. This is exactly what is found viz., curves A and B of Fig. 4 which are 1 in. and 1½ ins. respectively from the concave surface, also curve L of Fig. 5 which is 1 in. from the concave surface. As we move further from the concave to the convex surface the true secondary vorticity will become more important and should show an increase followed by a considerable decrease of α_2 from centre line to wall. This clearly is the flattening at about 2 ins. from the wall of the curves C to G of Fig. 4, each of which is progressively nearer the convex surface. The same effect is even more evident in curves M, N and P of Fig. 5. It will however be noticed that the last point of curve P is higher than the remainder. This may be due to the blade boundary layer separation in the corner and the refilling of the gap so created by higher energy air from the adjacent passage (see Ref. (5)), or to the reorientation predicted by the actuator plane.

At even higher negative incidences the increase in over turning of the air between the blade centre line and the wall boundary layer should become more evident.

If the streamlines near the leading edge could be accurately obtained there is no reason to suppose that the effects could not be calculated by using the theory of Hawthorne[6]. This however is not at present practicable.

Conclusion

Measurements have shown that when blades operate at high negative incidences in non-uniform streams a variation of outlet angle is obtained which may be qualitatively explained by considering the detailed curvature of the streamlines. The α_2 variation is considerably different from that anticipated using the conventional theory.

References/

References

No.	Author(s)	Title, etc.
1	W. R. Hawthorne	Rotational flow through cascades. Part I - The components of vorticity. Quarterly Journal of Mechanics and Applied Mathematics, Vol. VIII, Part 3, September, 1955, p. 266.
2	W. R. Hawthorne and W. D. Armstrong	Rotational flow through cascades. Part II - The circulation about the cascade. Quarterly Journal of Mechanics and Applied Mathematics, Vol. VIII, Part 3, September, 1955, p. 280.
3	W. D. Armstrong	The secondary flow in a cascade of turbine blades. R. & M. 2979. March, 1955.
4	W. R. Hawthorne and W. D. Armstrong	Shear flow through a cascade. To be published in Aeronautical Quarterly, November, 1956.
5	W. D. Armstrong	An experimental investigation of the secondary flow occurring in a compressor cascade. To be published in Aeronautical Quarterly.
6	W. R. Hawthorne	Secondary circulation in fluid flow. Proc. Roy. Soc. A. 206, 374.

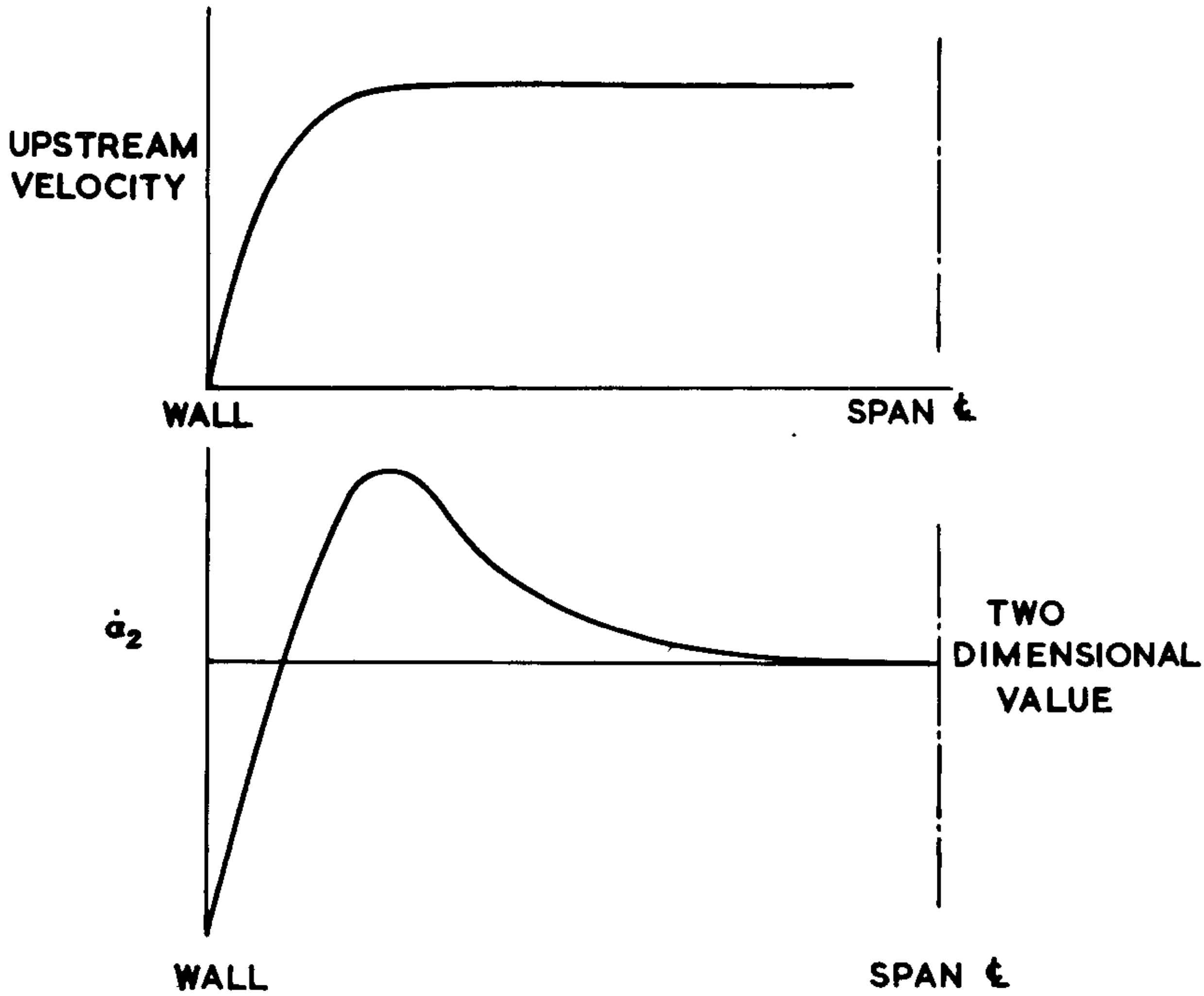

UPSTREAM
VELOCITY
WALL
SPAN ℄
$\dot{a}_2$
TWO
DIMENSIONAL
VALUE
WALL
SPAN ℄

FIG. 1.

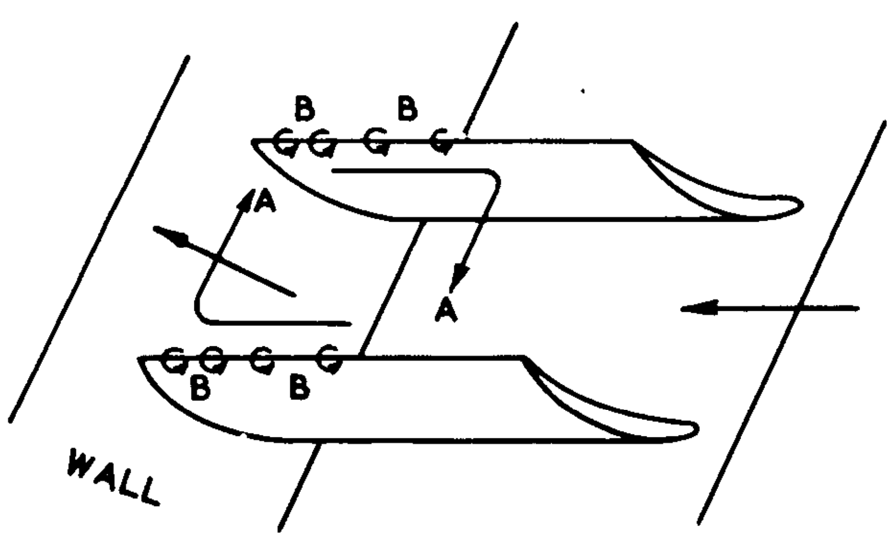
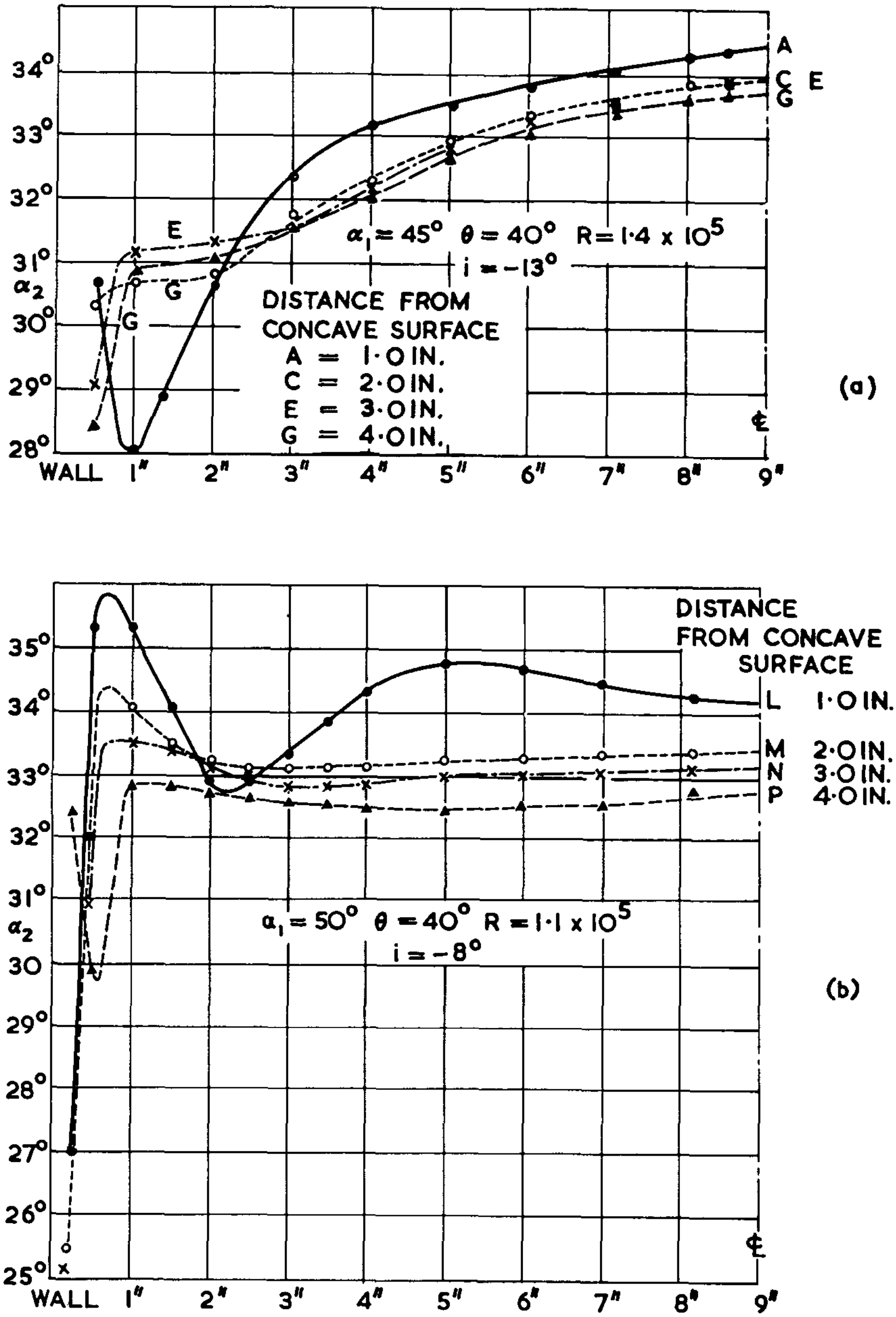

B
B
A
A
B
B
WALL

FIG. 2



34°
33°
32°
31°
α_2
30°
29°
28°
WALL 1" 2" 3" 4" 5" 6" 7" 8" 9"
A
C E
G
E
G
G
$\alpha_1 = 45°$ $\theta = 40°$ $R = 1{\cdot}4 \times 10^5$
$i = -13°$
DISTANCE FROM
CONCAVE SURFACE
A = 1·0 IN.
C = 2·0 IN.
E = 3·0 IN.
G = 4·0 IN.
℄
(a)
35°
34°
33°
32°
31°
α_2
30
29°
28°
27°
26°
25°
WALL 1" 2" 3" 4" 5" 6" 7" 8" 9"
DISTANCE
FROM CONCAVE
SURFACE
L 1·0 IN.
M 2·0 IN.
N 3·0 IN.
P 4·0 IN.
$\alpha_1 = 50°$ $\theta = 40°$ $R = 1{\cdot}1 \times 10^5$
$i = -8°$
℄
(b)

FIG. 3.

COMPRESSOR CASCADE

AIR OUTLET ANGLE α_2 (ONE CHORD DOWNSTREAM)

$\alpha_1 = 45°$ $\theta = 40°$ $R = 1{\cdot}4 \times 10^5$

NOTE: CURVES STAGGERED RELATIVE TO ONE ANOTHER BY ONE DEGREE

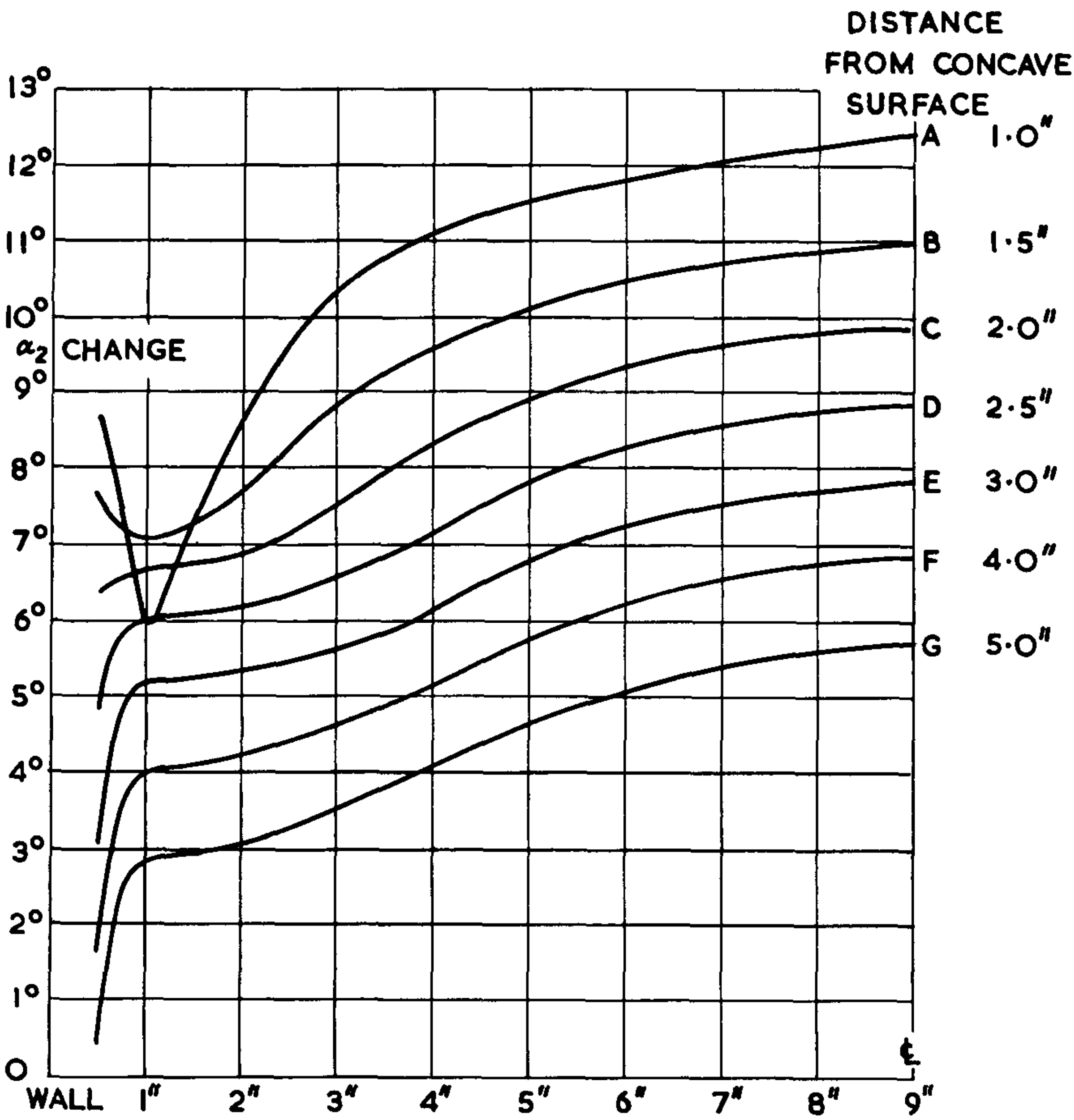


FIG. 4.

COMPRESSOR CASCADE

AIR OUTLET ANGLE α_2 (HALF CHORD DOWNSTREAM)

$\alpha_1 = 50°$ $\theta = 40°$ $R = 1{\cdot}1 \times 10^5$

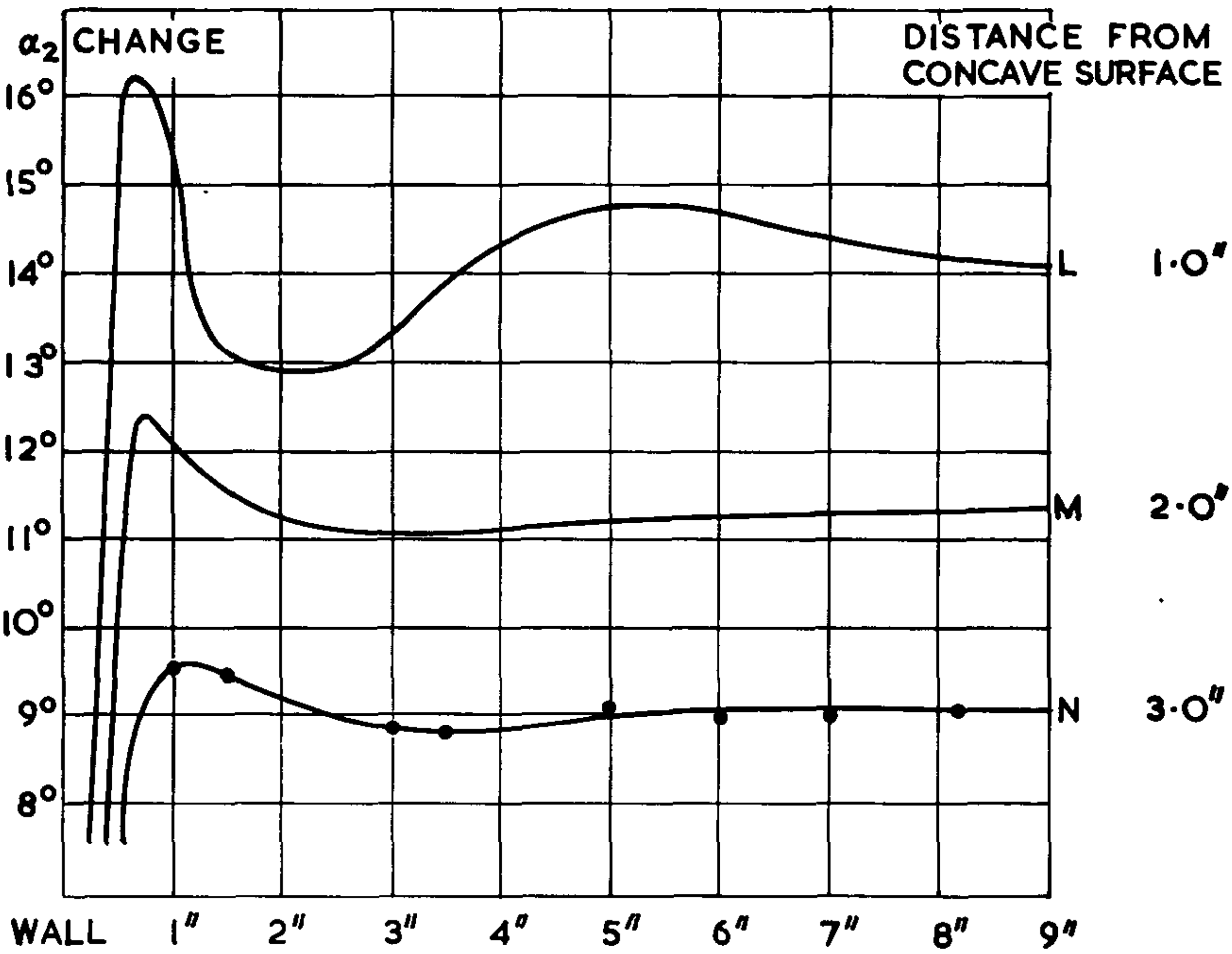


FIG. 5.

EFFECT OF REYNOLDS NUMBER AND INLET AIR ANGLE ON OUTLET AIR ANGLE

40° CAMBER CASCADE

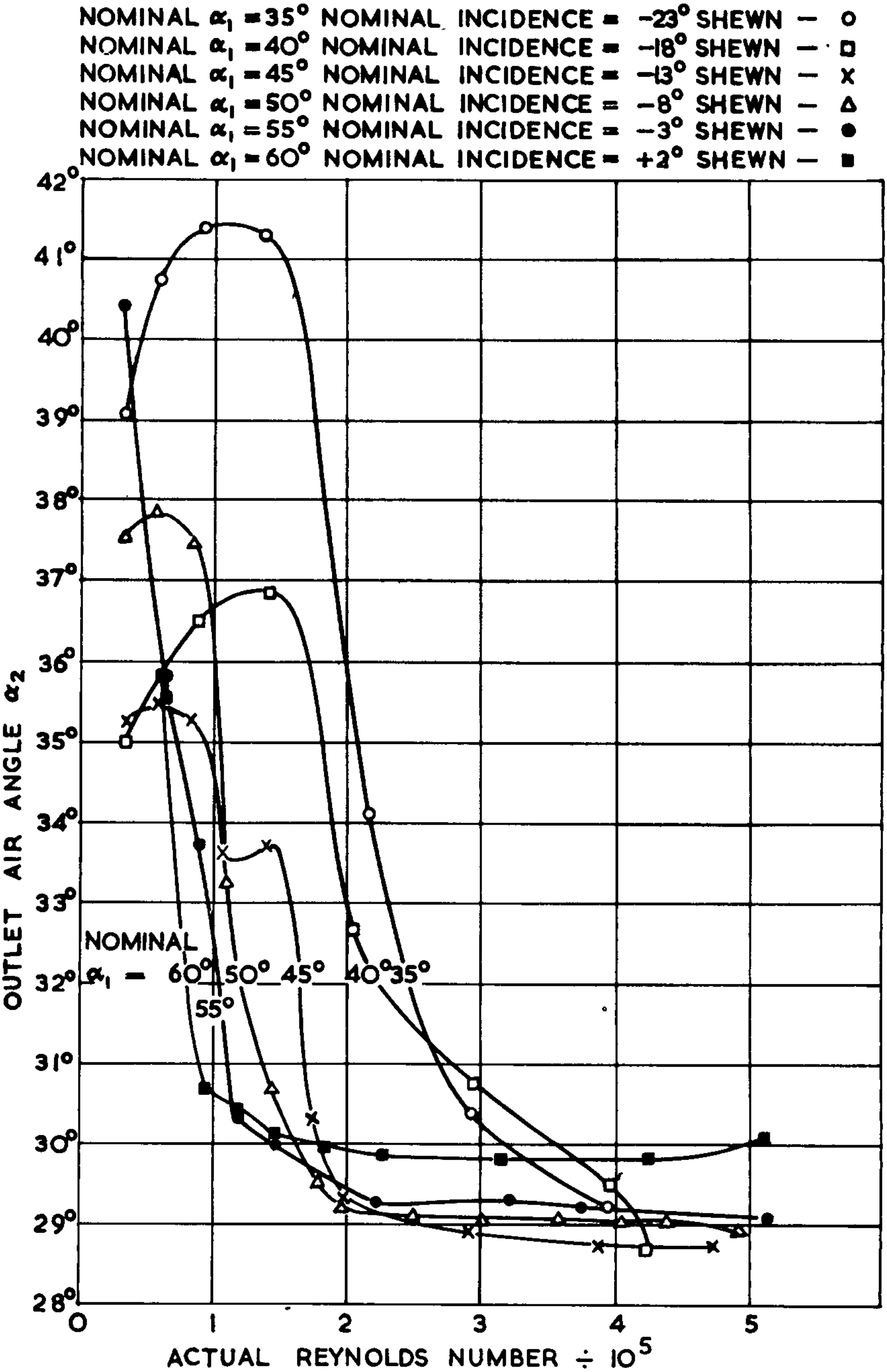


FIG. 6.

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